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FRICTION AND WINDAGE LOSSES IN ROTOR BLADE ASSEMBLIES OF RADIAL-FLOW TURBINES OF HELICOPTER AND TRANSPORT GAS TURBINE POWER PLANTS

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Friction and windage losses in blade rims of radial-flow turbines, at idling in forward and reverse direction and at variable rpm, are discussed. Results of experiments with Ljungström turbine blades mounted to the shaft of a high-frequency 9-kw motor in atmospheric air, in sealed and unsealed casings, are applied to aircraft gas turbines. Formulas are given for calculating the power expenditure at idling, allowing for ventilation of the medium and for boundary friction, in single- and multistage radial-flow turbines. Windage losses were reduced by modified layout of interblading shrouds without change in radial clearance.

The literature contains no investigations of power losses due to idle rotation of the blade rims of radial-flow turbines.

Taking into account the use of single-rotor and two-rotor radial-flow turbines in helicopter gas turbine plants (see, for example, British Patent 820337, Class 4, 1956), where the magnitude of these losses affects the landing speed of a helicopter with an idling gas turbine and determines the need for a free-wheeling clutch, and their use in transport aircraft power plants requiring reversal from the power turbine, we carried out investigations which to a certain extent should fill this gap.

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* Numbers in the margin indicate pagination in the original foreign text.

One of the objects of the investigation was the blading taken from the Ljungström radial-flow multistage turbine. As is known, the design of the blading of the Ljungström radial turbines provides for high thermoelasticity and the capacity of almost constant maintenance of the size of the interblading radial clearance during rapid and appreciable changes in operating conditions and at excessive heating of its components. For example, the aircraft two-rotor radial-flow gas turbine of the Ljungström system successfully operated in 1935 at an initial gas temperature of 923 - 973°K (Bibl.1). Thanks to these properties, the design of the Ljungström blading is most suitable for turbines of which one expects high maneuverability, reliability, and economy, and also for applying, under multistage turbine conditions (absence of vacuum), well-known methods for reducing friction and windage losses in the working rims at protracted idling such as the use of annular casings and reduction in density of the medium (by appreciably raising the temperature of the medium by using the heat generated by the friction losses).

1. Conditions, Means, and Methods of the Experiments

The investigations were carried out on a model and experimental device whose diagram (for testing radial blade rings) is given in Fig.1a. The blade rings (1) of the Ljungström turbine, having 208 blades of 7.7 mm width and 18 mm length with an outside diameter of 366.4 mm, was mounted directly to the shaft of a high-frequency 9-kw electric motor (2) rotating at 8000 rpm in a medium of atmospheric air inside an enclosure (10), in forward and reverse /61 directions at variable rpm under the following conditions:

1) in free space, without any blading elements;

2) in the presence of guide blading (8) coupled with it (Fig.1b);

3) in annular space between simply supported rings of the guide-vane rims (Fig.1c).

Axial displacement of the concentric blade rings (8) from the position b to c occurred both during rotation and upon stopping the row of blades without

Section through A-A

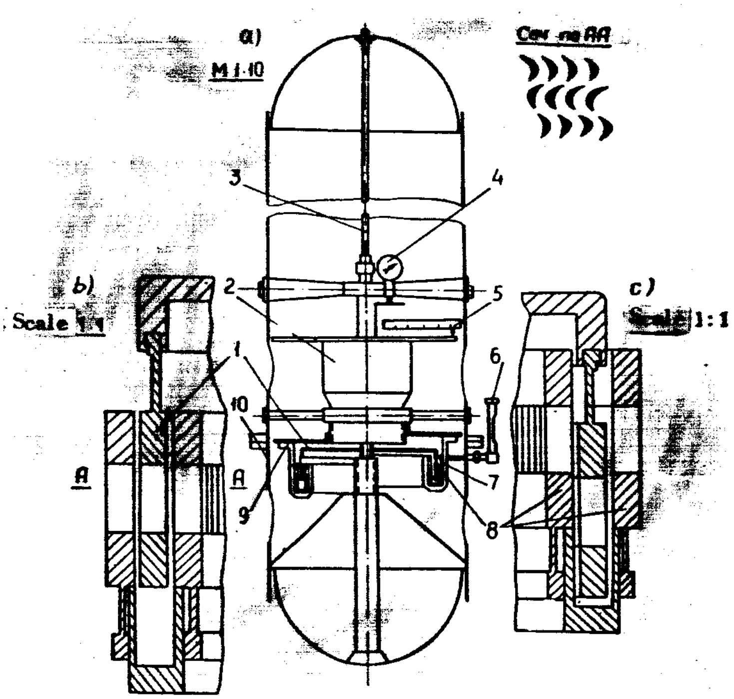

Fig.1 Cut-Away of the Device
1 - Moving blades; 2 - Electric motor; 3 - Torsional shaft; 4 - Tachometer; 5 - Dial and pointer; 6 - Thermometer; 7 - Angle bracket; 8 - Guide vanes; 9 - Supporting disk; 10 - Casing

Scale

catching between the blades, and was accomplished by turning the supporting disk (9) holding the blade rings by three angle brackets (7); the disk was attached by a threaded connection to the casing of the lower bearing crosspiece of the electric motor, coaxially with its shaft. The axial displacement of the blade rings in motion was accomplished in an analogous manner as the displacement of the exit disks of the first Ljungström radial-flow turbine, in whose labyrinth seals the clearances were considerably smaller than in the case under consideration (Bibl.2). The rpm was measured by a mechanical tachometer (4) with an accuracy of up to 5%; the temperature of the working medium was determined by a mercury thermometer close to the blade rims with an accuracy of 0.5°, while the barometric pressure was taken from the data of the Meteorological Service. The moment of forces acting on the radial rim, equal to the shaft torque of the electric motor, was determined by the reaction moment on the stator measured in the device by the angle of twist of the torsional shaft (3) to which the stator was mounted, forming a torsion balance with it. The /62 angle of twist was measured visually from a fixed dial and a pointer (5) attached to the stator. The torsiometric system was calibrated in the device by a spring-filled dynamometer and a balance, with the electric motor operating and stopped. Calibration was performed before and after the experiments. The results of numerous calibrations during the year coincided, when the dynamometer and balance were put in use. The power lost to rotation of the radial blade rings was determined by the formula

$$N = C \cdot \frac{\varphi \cdot L}{75} \cdot \frac{\pi \cdot n}{30} = \frac{L}{16.7} \cdot \frac{\varphi \cdot n}{1000}.$$

where

N = power, kw;

$C = \frac{G}{\varphi} \cdot g$ = characteristic of the torsiometric system, H/mm;

G = calibration force, H;

φ = length of arc on scale corresponding to angle of twist, mm;

L = calibration lever arm, m;

n = rotational speed, rpm;

g = acceleration of gravity, m/sec^2.

To plot the curves of the power loss due to friction and windage as a function of the rotatory speed, we found it more convenient to use the quantity

$\frac{\varphi \cdot n}{1000}$, which is proportional to the power and is reduced to the following conditions: air pressure 1.013 bar or 760 mm Hg and temperature of 293°K. The total error of determining the power in the range of low rpm was about 9%. The reduction was performed by the formula

$$\left(\frac{\varphi \cdot n}{1000}\right)_{red} = \frac{\varphi \cdot n}{1000} \cdot \frac{760}{B} \cdot \frac{T}{293},$$

where

B = barometric pressure, mm Hg;

T = temperature of the medium, °K.

2. Experimental Results

The results of the experiments are presented in Fig.2, in a logarithmic coordinate system.

Curves 1 and 5 correspond to idle rotation of the working rim in reverse and forward directions between the blade rings aligned in the position shown in Fig.1b.

Curves 2 and 4 characterize the magnitude of the friction and windage losses in a single blade ring, rotating respectively in reverse and forward directions in free space.

Curves 3 and 6 give the magnitude of these losses during reverse and forward rotation of the blade ring in the annular clearance between the supporting rings of the guide-vane rims in the position shown in Fig.1c. A comparison of the ordinate axes of curves 1 and 5, 2 and 4, and 3 and 6 shows that, if the direction of idling of the blade rings changes from forward to reverse and if other conditions are kept equal, the friction and windage losses will increase: in the first case (intermediate stage) by a factor of 15 - 20, in the second /63 (single blade ring) by 9 - 10, and in the third (blade ring in the annular casing) by 2 - 2.5.

As shown by the slope of the curves, in all of the above cases, at forward idle rotation of the blade rings, the magnitude of the friction and windage losses is proportional to the rpm with an exponent of 2.85 - 2.9 and, at reverse rotation, with an exponent of 3 - 3.3; in the given coordinate system, this is equal to the tangent of the angle of slope of the curves to the abscissa axis.

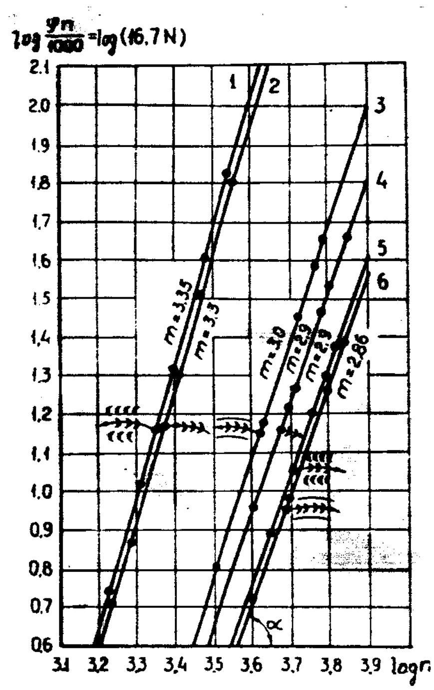

Fig.2 Results of Testing the Blade Rings of the Ljungström Radial-Flow Reaction Turbine, m = tan α

If, as a comparison standard, we take the level of the friction and windage losses in a single blade ring rotating in free space, then the effect of the structural elements on the magnitude of these losses in the blade ring can be described by the following relations:

At forward rotation of the blade ring, as indicated by curves 4 and 5, the guide-vane rims reduce the friction and windage losses by a factor of about 1.5, which is explained by the decrease in the rate of flow of the medium ventilated by the blade ring in the presence of guides, in comparison with the case of similar rotation of the blade ring in the absence of guides and in free space.

Bilateral housing of the blades of the working rim by simply supported rings of the guide rims, as shown by curves 4 and 6, leads to a drop in these losses by a factor of 1.6 - 1.7 in the case of forward rotation in comparison with the case of the same rotation of the working rim in free space, and by a factor of 1.1 in comparison with the case of its forward rotation under conditions of an intermediate stage, which is also explained by a decrease of ventilation of the working medium.

In the case of reverse idle rotation of the working rim, the presence of guide rims, as is apparent from a comparison of the ordinates of curves 1 and 2, causes an increase in friction and windage losses by a factor of 1.1 with respect to a single blade ring rotating in reverse in free space.

It follows from a comparison of the ordinate axes of curves 2 and 3 and of 1 and 3 that housing the blades of the working rim with reverse rotation by means of supported rings of the guide rims of the adjacent stages forming the annular space or by a special annular casing makes it possible to reduce friction and windage losses by a factor of 8 - 9 in comparison with the case of reverse rotation of a single blade ring in free space and by a factor of 9 - 10 in comparison with the case of the same rotation under conditions of an intermediate stage. The obtained results satisfactorily agree with the results of investigations of other radial rims and with the special experiments which /64 were previously published in part (Bibl.3, 4); this permitted establishing the quantitative relations between the power lost to ventilation of the medium and to overcoming the friction of the blade edges and, consequently, defining the principle and significance of the losses produced on interaction of the blades with the medium under free-space conditions and in annular casings. Visual observation and measurement of the flow angles of the medium at the entrance and exit of the radial rim, rotating in free space, showed that the rim interacts with the medium mainly like the working rim of a simplest radial-type blower.

Theoretical calculations, taking into account the ventilation of the medium and boundary friction, which were confirmed by experimental investigations (Bibl.3), gave the following formula for determining the friction and windage power in a radial rim:

$$N = k \cdot d^4 \cdot l \left(\frac{n}{1000}\right)^m \cdot \rho.$$

In groups of radial stages with blades of the same shape and comparatively small difference in lengths, this formula can be written in the form of

$$N = k \cdot g l_{cp} \cdot \rho_{cp} \cdot \left(\frac{n}{1000}\right)^m \cdot \sum_1^z d_i^4 .$$

In the case of a large number of uniform stages in a group with the same width of the rotor and stator blading and size of the inter-blade clearance, the sum of the fourth power of the outer diameters of the working rims can be approximately determined by integration:

$$\sum_1^z d_i^4 = \sum_1^z [d_1 + 4b(z-1)]^4 \simeq \int_1^z [d_1 + 4b(z-1)]^4 dz = \frac{[d_1 + 4b(z-1)]^5 - d_1^5}{20 \cdot b},$$

$$\sum_1^z d_i^4 \simeq \frac{d_z^5 - d_1^5}{20 \cdot b}.$$

Here, the following symbols are used:

N = friction and windage power, kw;

n = number of revolutions, rpm;

m = exponent on the rotational speed;

l, $l_{av} = \dfrac{l_1 + l_z}{2}$ = respectively, length and average length of the moving blades, cm;

d = outer diameter of the working rim, m;

ρ, $\rho_{av} = \frac{\rho_1 + \rho_z}{2}$ = respectively, density and average density of the medium, kg/m^3;

b = reduced width of the rims equal to the sum of the rim width and radial clearance, m;

z = number of stages;

k = coefficient taking into account the effect of the direction of rotation, geometric angles, blade width and pitch, structural elements, and size of the radial clearance.

The indices 1 and z denote the quantities pertaining to the first and last stages of the group. /65

According to the experiments with radial blade rings, the exponent for the rpm in all cases with forward idling can be taken as equal to 2.85 - 2.9 (rounded to 2.9) and with reverse rotation as 3 - 3.3 (rounded to 3).

Depending on the circumstances, the coefficient K on the basis of the results of testing the Ljungström radial-flow blade rings can be taken as follows:

TABLE

	Forward Rotation	Reverse Rotation
1. Rim in free medium	0.257	2.57
2. Rim in intermediate stage	0.162	2.94
3. Rim in annular casing or between the supporting rings of the guide rims in the absence of interblading labyrinth seals	0.147	0.294

With interblading labyrinth seals in which the clearances are always smaller than the clearances between the rims, there is an additional decrease in the windage losses in the third case by a factor of 2 - 2.5 (Bibl.4), produced by the decrease in ventilation of the medium.

The possibility of an additional substantial decrease in windage losses in the third case, by organizing the shrouds while retaining sufficiently large clearances between the rotating and fixed massive parts, is of considerable practical interest since simultaneously with a reduction in losses, the reliability of the turbine is increased: Possible interference of comparatively thin elements of the shrouds upon marked changes of conditions cannot lead to an accident or to appreciable changes in the friction and windage losses since the size of the clearance in the interblading shrouds of existing radial-flow turbines is 5 - 10 times smaller than the clearance between the rims (Bibl.5).

Limitations on the length of this article do not permit a detailed account of all the experiments performed; therefore, we will restrict ourselves to listing the basic results of the work carried out on friction and windage losses in radial bladings.

1. Friction and windage losses in radial blade rings during idling in forward and reverse directions with a variable rpm were investigated:

a) in free space, b) in annular labyrinth sealed and unsealed casings, c) under conditions of a single centrifugal stage, d) under conditions of an intermediate stage of a multistage turbine, e) in the annular space between simply supported rings of the guide rims of adjacent stages of multistage turbines.

2. The quantitative relationship between power expended on ventilation of the medium and on friction of the blade edges was determined; the essence and significance of the losses during idle rotation of radial blade rings in free space and in annular casings were elicited.

3. The effect on the magnitude of the friction and windage power expended for idle rotation of the radial rim, of the following factors was established:

a) direction of rotation, b) rpm, c) length of the moving blades, d) width of the working rims, e) exit angle of the nozzle blades in a single centri- /66 fugal stage, f) structural elements of the blading, g) interblading labyrinth seals and radial clearance with rotation in annular casings and in the annular space between the simply supported rings of the guide vanes.

4. An experimental comparison was made of the friction and windage losses in radial-flow and axial-flow working blade rings.

5. Design methods of reducing friction and windage losses in radial rims were discussed.

6. Formulas and coefficients were proposed for practical use, which permit calculating the power expended for idle rotation of radial-flow blade rings in single-stage and multistage radial-flow turbines.

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